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THE DIESEL AS A VEHICLE ENGINE

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The economical distress of the post-bellum years led the German Diesel-engine builders to take up anew the task of producing a high-speed vehicle engine. Their endeavors were given a mighty stimulus by the rapid development of the compressorless engines, for it is obvious that only this type of Diesel engine can be used on vehicles.

The thorough investigation of a Dorner four-cylinder, four-stroke-cycle Diesel engine with mechanical injection in the Institute for Internal Combustion Engines of the Hannover Technical High School led me to investigate more thoroughly the operation of the Diesel as a vehicle engine, in order to get an idea of the development possibilities of this type of engine.

Aside from the obvious need of reliability of functioning, a high rotative speed, light weight and economy in heat consumption per horsepower are also indispensable requirements. The present commanding position of gasoline carburetor engines is due to their fulfillment of these conditions. All attempts to use heavy oils with carburetors have met with failure. The

*"Die Dieselmaschine als Kraftfahrzeugmotor." From Zeitschrift des Vereines Deutscher Ingenieure (V.D.I.), May 28, 1927, pp. 775-785.
For previous articles on "Experiments with Diesel Engines," see V.D.I. 1918, p.706; 1921, p.801; Forschungsheft 245; 1923, pp.279 and 755; 1924, p.283; 1926, p.1071 (N.A.C.A. Technical Memorandum No. 391).

first turning point came with the introduction of the Diesel process.

Strange to say, the difficulties of the Diesel vehicle engine were not found in the direction they were formerly supposed to be, before the compressorless engine had passed its initial stages. Very small quantities of fuel, corresponding exactly to the load, can now be delivered to each cylinder and be completely consumed. As will be shown later by the experimental results, this is the basis for the assumption of efficient utilization of the heat energy of heavy oils in high-speed engines.

The high rotative speeds require that the fuel injected into the combustion space shall undergo the necessary changes in condition and be consumed with maximum rapidity. The practicability of the process depends on the relation of this speed to the piston speed in every phase of the process, and especially on the pressure curve during the combustion and expansion in the cylinder, and consequently, on the maximum pressure, which is decisive for the strength calculations and the weight of the engine. The combustion can be rapid and complete, however, only when the fuel is brought into contact with an adequate supply of air. The highest possible cylinder efficiency with relation to the indrawn air must therefore be sought, even at high rotative speeds.

Working Process of Diesel Vehicle Engine

This will first be considered on the basis of the theoretical indicator diagram (Fig. 1), according to which the combustion takes place partially at constant volume and partially at constant pressure. Since we are now concerned only with the determination of comparative values, the specific heats are considered constant, and thermal dissociation of the combustion products is disregarded. Compression and expansion are assumed to be adiabatic.

The engine is supposed to function at a constant compression ratio. Of each weight unit of the mixture of fuel and air, x denotes the portion burned at constant volume and $1 - x$ the portion burned at constant pressure. The value of x varies between 0 and 1.

The condition of the air at the end of the intake is $p_1 = 1$ atm. abs., $t_1 = 80^\circ C$ and the final compression pressure is $p_2 = 25$ atm. abs. The fuel is a heavy oil, which has a heating value of $h_u = 8500$ kcal/kg and for whose combustion the air quantity $L_{min} = 13.85$ kg is theoretically required for 1 kg of fuel. It is burned with 70% excess air, so that $\lambda = 1.70$. Furthermore let

$$c_p = 0.24, \quad c_v = 0.1715 \left[\frac{kcal}{kg^\circ C}\right], \quad k = \frac{c_p}{c_v} = 1.4$$

Then the compression ratio of the engine is

$$\epsilon = \frac{v_1}{v_2} = [\frac{p_2}{p_1}]^{\frac{1}{k}} = 9.91,$$

and the final compression temperature is

$$T_2 = T_1 \epsilon^{k-1} = 886 [^\circ \text{abs.}], \quad t_2 = 613 [^\circ C].$$

The relative pressure and volume increases from the combustion depend on the amount of fuel burned.

$$\text{For } V_k = \text{constant}, \quad \psi = \frac{p_3}{p_2} = f(x)$$

$$\text{" } P_3 = \text{ " }, \quad \varphi = \frac{v_4}{v_3} = f(1 - x).$$

The temperatures at the close of the partial combustions are

$$T_3 = T_2 + \frac{x\ h_u}{c_v(1+\lambda L_{min})}, \quad T_4 = T_3 + \frac{(1-x)h_u}{c_p(1+\lambda L_{min})} [^\circ \text{abs.}]$$

and herewith

$$\psi = 1 + \frac{x\ h_u}{c_v\ T_2(1+\lambda L_{min})}, \quad \varphi = 1 + \frac{(1-x)h_u}{k\ x\ h_u + c_p\ T_2(1+\lambda L_{min})}$$

whereby p_3 and v_4 are determined. The engine charge is

$$\xi = \frac{v_4 - v_3}{v_1 - v_2} = \frac{\varphi - 1}{8.91}.$$

Since the total heat

$$Q = \frac{h_u}{1 + \lambda\ L_{min}} = 347 \text{ [kcal]}$$

freed by the combustion of 1 kg of the mixture is always the same and 1 kg of the mixture does the work

$$A\,L = A\,P_3\,(v_4 - v_3) + (u_4 - u_5) - (u_2 - u_1)$$

$$= 0.0685\;(t_4 - t_3) + c_v\,[(t_1 + t_4) - (t_2 + t_5)]\;[\text{kcal}]$$

the efficiency of the engine is

$$\eta = \frac{A\,L}{Q}$$

The mean indicated piston pressure is

$$p_i = \frac{10^{-4}\,L}{v_1 - v_2}\,[\text{at}].$$

Table I and Figure 2 show the results of the calculation. In Figures 3-4 the corresponding indicator and T_S diagrams are quantitatively plotted for 1 kg.

Table I.

Effect of Preignition on the Working Process

x kg	0	0.2	0.4	0.6	0.8	1.0
ψ	1	1.453	1.906	2.36	2.81	3.26
φ	2.613	1.887	1.509	1.273	1.114	1
ξ . . . . . %	18.1	9.95	5.70	3.07	1.28	0
p_3 at abs.	25	36.3	47.6	59	70	81.5
p_5 " "	3.85	3.54	3.40	3.30	3.27	3.26
t_3 . . . . °C	613	1012	1417	1822	2217	2627
t_4 . . . . . "	2037	2157	2277	2387	2487	2627
t_5 . . . . . "	1077	971	925	897	883	882
AL . . kcal	170.9	190.4	199.2	201.5	202.7	207.5
pi . . . at	7.29	8.13	8.51	8.65	8.76	8.85
η . . . . . %	49.2	55	57.5	58.1	58.5	59.9

The experiments show that a given quantity of fuel with a constant quantity of air, in an engine with constant final compression pressure, burns the best at constant volume, though the maximum pressure is then very high. The increase in the mean indicated pressure p_i, in the transition from combustion at constant pressure ($x = 0$, $p_{max} = 25$ atm. abs.) to combustion at constant volume ($x = 1$, $p_{max} = 81.5$ atm. abs.), is 21.6%. It is noteworthy, however, that this increase is very slight for $x > 0.5$, so that, even with slight preignition, the energy is favorably utilized at moderate maximum pressures.

If, for example, half of the injected fuel is burned at constant volume and the other half at constant pressure, then, with $x = 0.5$,

$$p_{max} = 53.1 \text{ [at abs.]} \qquad t_{max} = 2330 [^\circ C]$$

$$\xi = 4.2\,\% \qquad t_5 = 911 [^\circ C]$$

$$p_i = 8.58 \text{ [at]} \qquad \eta = 57.9\%$$

i.e., p_i and η remain only a little below the attainable maximum values, although p_{max} is only about 2/3 of the highest possible pressure. The actual pressure curve in the cylinder of a compressorless Diesel engine during the combustion depends on the rapidity of transformation of the injected fuel and therefore always requires a finite time. The higher the rotative speed of the engine is, the more the pressure curve will deviate from the curve of constant volume, even with very rapid injection

and combustion. Accordingly, it is fundamentally wrong arbitrarily to assume the maximum pressure of the theoretical indicator diagram and consequently to resolve the combustion into the two parts V = constant and P = constant,* because the combustion speed is thus assumed without taking into account the physical and chemical conditions on which the combustion alone depends. The efficiency of the working process, thus calculated, is uncertain and affords no criterion for the perfection of the working process of the engine.

In order to gain an insight into the rapidity of the combustion of heavy oils and its dependence on the different conditions, we must also undertake special experiments which will enable the systematic variation of these conditions. For this purpose a new series of experiments has been undertaken in the combustion-engine laboratory of the Hannover Technical High School, in which the previously described apparatus will be used to some extent (See N.A.C.A. Technical Memorandum No. 391).

a) Combustion at constant volume.- The time-pressure curve of the combustion within a constant space can, with the exclusion of heat losses on the basis of experimental results, be approximately represented by the exponential function $p = Ae^{Bz}$ atm. abs., in which the constants are determined by the conditions that $z = 0$ for $p = p_2$ and $z = z_V$ for $p = p_{max}$.

*Maier, "Untersuchung einer Deutzer VM-Maschine" in V.D.I. 1924, p. 1052.

p_2 denotes the pressure in the chamber before the combustion, p_{max} denotes the maximum pressure produced by the assumed perfect combustion, and z_v denotes the time required for the reaction from the beginning of the pressure increase. It is

$$p = p_2 \, e^{\ln\left(\frac{p_{max}}{p_2}\right)\frac{z}{z_v}} \text{ [at abs.]}$$

The temperature corresponding to the pressure p is

$$T = \frac{T_2}{1 - \dfrac{\left(1 - e^{-\ln\left(\frac{p_{max}}{p_2}\right)\frac{z}{z_v}}\right)}{1 + \dfrac{\left(\frac{c}{12} + \frac{h}{2} + \frac{w}{18} - \frac{0.21\ L_{min}}{24.4}\right)[\mu\ c_v'']_{T_2}^{T}\ T_2}{h_u}}} \text{ [}^{\circ}\text{abs.]}$$

and the portion of the injected fuel, consumed up to that time, is

$$x = \frac{\dfrac{\lambda\, L_{min}}{24.4}}{\dfrac{h_u}{[\mu\ c_v'']_{T_2}^{T}\ T_2} + \left(\frac{c}{12} + \frac{h}{2} + \frac{w}{18} - \frac{0.21\ L_{min}}{24.4}\right)} \times \left(e^{\ln\left(\frac{p_{max}}{p_2}\right)\frac{z}{z_v}} - 1\right) \text{ [kg].}$$

in which c, h and w respectively, represent the carbon, hydrogen and water contained in the fuel. The rapidity of the reaction at the point p or T at the time z is then obtained by differentiation.

$$\frac{dx}{dz} = \frac{\frac{\lambda\, L_{min}}{24.4\; z_v} \ln \frac{p_{max}}{p_2}}{\frac{h_u}{[\mu\; c_v'']_{T_2}^{T}\; T_2} + \left(\frac{c}{12} + \frac{h}{2} + \frac{w}{18} - \frac{0.21\; L_{min}}{24.4}\right)} \times e^{\ln\left(\frac{p_{max}}{p_2}\right)\frac{z}{z_v}} [kg/s] \; .$$

In all cases the maximum pressure and temperature of the combustion, at V = constant, are

$$p_{max} = p_2\; \frac{m''}{m_2}\; \frac{T_{max}}{T_2}\; [\text{at abs.}]$$

and

$$T_{max} = T_2 + \frac{h_u}{m''\; [\mu\; c_v'']_{T_2}^{T_{max}}} [^\circ \text{abs.}]$$

in which m_2 and m'' denote the molecular weights of the enclosed air or combustion products.

b) Calculation of the combustion curve with increasing cylinder volume.- In order to calculate the working process of a high-speed Diesel vehicle engine, neglecting heat losses by radiation and conduction, it is now assumed that the fuel is injected into the cylinder so early that the pressure increase due to the combustion begins at the firing dead center of the piston. With stationary piston the maximum pressure, after the time $z = z_v$, would be $p = p_{max}$ atm. abs. With a forward-moving piston, the cylinder volume has increased, in the time z, to $v = v_k + \frac{v_h}{2} (1 - \cos \alpha)$ [m³] indicating that an adiabatic expansion to the pressure $p < p_{max}$ has simultaneously occurred.

The final compression pressure p_4 (Fig. 5) is hereby all the smaller than p_{max} for v_k = constant, the farther the piston has moved from the dead center. In order to find the real combustion curve in the p V diagram, we must calculate the constantly changing condition constants p and T from the computed time curve of the chemical transformation of the fuel into heat $x = f(z)$. For this purpose, the whole combustion process 2-4 is resolved into elementary processes and the actual combustion curve is calculated step by step, in that, instead of the actual combustion corresponding to the section a b of the curve, a substitution process a to c with v = constant and a second c to b at T = constant, is carried out in such manner that the external work of both processes differs only by an infinitely small quantity of the second order of magnitude.

For such a partial process, the first law gives

$$d\,U + h_u\,dx = AP\,d\,V,$$

$$U_a + (x_b - x_a)\,h_u = U_b + A\int_a^b P\,d\,V$$

or

$$m_a \int_0^{t_a} \mu\,c_v''\,dt + (x_b - x_a)\,h_u$$

$$= m_b \left\{\int_0^{t_b} \mu\,c_v''\,dt + 1.985\,T_b\,\ln \frac{1 + 2\,\epsilon_o - \cos\alpha_b}{1 + 2\,\epsilon_o - \cos\alpha_a}\right\}$$

whereby

$$m_n = \left(\frac{c}{12} + \frac{h}{2} + \frac{w}{18} - \frac{0.21\,L_{min}}{24.4}\right)x_n + \frac{\lambda\,L_{min}}{24.4}\left[\frac{\text{Mol.}}{\text{kg fuel}}\right]$$

and

$$\epsilon_o = \frac{V_k}{V_h}$$

is the relative volume of the compression space in the engine (Cf. V.D.I. 1923, p. 280).

With a known initial condition a and a known fuel transformation $x_b - x_a$ corresponding to the time increase $z_b - z_a$ or an increase of $\alpha_b - \alpha_a$ in the crank angle, we can next determine the temperature in the final condition b and then, from the equation

$$p_b V_b = 848 m_b T_b,$$

the corresponding pressure p_b.

Proceeding from condition 2 at the end of the compression, in which

$$\alpha_a = 0, \quad x_a = 0, \quad t_a = t_2, \quad m_a = m_2$$

the calculation leads step by step to the final condition $x = 1$, $m_b = m''$, p_4, V_4, t_4 corresponding to the assumed combustion period z_v (sec.) or to the corresponding crank angle $\alpha_v = 6n\, z_v$ (degrees), in which n denotes the R.P.M. of the engine.

From the above-developed formulas, without first considering a special numerical example, we can draw a series of important conclusions.

The rapidity of the pressure increase during combustion depends on the constant

$$B = \frac{1}{z_v} \ln \frac{p_{max}}{p_2} \, [s^{-1}]$$

In the first place, the combustion period z_v or the combustion speed (which is inversely proportional to the combustion

period) has an influence; and, in the second place, the pressure increase due to the combustion.

The combustion period depends on the temperature, density of air in combustion chamber, concentration, heat conductivity and absorption capacity of the burning fuel mixture, vaporization speed of the fuel and, above all, on the turbulence of the mixture. The pressure increase, on the contrary, is, in the main, a function of the excess air factor λ. It is greatest when $\lambda = 1$. It can be diminished by dissociation of the combustion products. The law of gravity also plays an important role in the form to be used for kinetic reaction phenomena.

Obviously extensive experimental and mathematical researches will be necessary, in order to explain the phenomena in the combustion of heavy oils and to discover the relations which govern the time element of the chemical reactions.

Since the rotative speed n is very high in a vehicle engine, the transformation speed dx/dz must be very great for the rapid conversion of the fuel into heat and mechanical energy, which can be accomplished, as shown by the formula, only by a short duration z_v of the combustion, i.e., by a high combustion speed. It is also obvious that a relatively high temperature T_2 at the end of the compression has a similar effect. A high compression not only diminishes the ignition delay, but also increases the combustion speed (N.A.C.A. Technical Memorandum No. 391). The transformation speed $\frac{dx}{dz} = f(z)$ is not constant dur-

ing the combustion, but increases approximately according to an exponential function

$$e^{\ln\left(\frac{p_{max}}{p_2}\right)\frac{z}{z_v}}$$

For $z = z_v$ at the end of the combustion, it is more than p_{max}/p_2 times as large as for $z = 0$ at the beginning of the combustion. A less excess of air (large p_{max}) is advantageous, so long as the combustion can still be regarded as complete.

The expression $p = Ae^{Bz}$, which characterizes the pressure curve at constant volume was adopted as the basis of the experiments with reference to the provisionally established facts. It is the task of further research to trace the combustion period z_v of the heavy oils to its physico-chemical causes and represent it as a function of the abovementioned individual characteristic quantities. There is then no difficulty in proceeding by the designated way from the combustion at constant volume to combustion at increasing cylinder volume and thereby establishing, in particular, the effect of the rotative speed or piston speed on the efficiency of the energy transformation.

c) Example and conclusions.- A heavy oil (gas oil: $c = 0.87$; $h = 0.13$ kg; $h_u = 10,100$ kcal/kg; $L_{min} = 12.21$ m³/kg at 15°C and 760 mm Hg) is burned with $\lambda = 1.50$ excess air in a compressorless mechanical injection engine, neglecting heat losses by radiation and conduction. At the beginning of

the compression $p_1 = 1$ atm. abs. and $t_1 = 57^\circ C$. At the end, $p_2 = 30$ atm. abs. and $t_2 = 598^\circ C$. Based on 1 kg of fuel,

$$m_2 = \frac{\lambda\, L_{min}}{24.4} = 0.751 \text{ mol. air,}$$

$$m'' = 0.0725\ CO_2'' + 0.0650\ H_2O'' + 0.0526\ O_2'' + 0.594\ N_2''$$

$$= 0.784 \text{ mol. combustion products.}$$

Burned at constant volume, we obtain, for the final condition of the combustion, with consideration of the variability of the mean molecular heat (Hütte, Vol. I, edition 25, p. 472),

$t_{max} = 2450^\circ C$ and $p_{max} = 97.8$ atm. abs. The variation of the characteristic quantities for fractions of the combustion period z_v is shown in Table II and Figure 6.

Table II

Variation of Characteristic Quantities for Combustion at Constant Volume

$\frac{z}{z_v}$	0	0.25	0.50	0.75	1
p .... at abs.	30	40.4	54.1	72.9	97.8
t .... °C	598	870	1242	1740	2450
x .... kg	0	0.130	0.313	0.559	1
$\frac{dx}{dz}$ .... kg/s	73.9	117.9	166.2	236.8	343.3
m_x'' .... mol/kg	0.751	0.755	0.761	0.769	0.784
$[\mu\, c_v'']_0^t$ kcal/mol °C	5.11	5.32	5.61	6.01	6.54

The law for the conversion of the chemical energy of the fuel into heat $x = f(z)$ is now regarded as valid for the combustion in the engine with increasing cylinder volume, so that the combustion ends at various crank angles α_v. If n denotes the rotative speed of the engine, then the combustion period $z_v = \frac{\alpha_v}{6n}$ sec. is determined for each individual case. It is obvious that constantly increasing combustion periods z_v or constantly decreasing fuel speeds correspond to the increasing crank angles α_v, since there is no change in the final condition of the compression (p_2, t_2) nor in the rotative speed. The combustion period $z_v = 0$, for which the combustion speed is infinitely great, corresponds to the crank angle $\alpha_v = 0$. Since the combustion speed, however, can have only finite values, it follows that the combustion and the corresponding maximum pressure p_{max} at $V =$ constant can have only the status of limiting values, even for engines with heat losses neglected.

The variation of the characteristic quantities for different parts of the variable combustion period z_v for $n = 1000$ R.P.M. is shown in Table III and Figure 7, while Figure 8 contains the corresponding indicator diagrams. We recognize the great effect which the combustion speed, under otherwise similar conditions, has on the indicated horsepower of the engine. The lower the combustion speed, the lower the maximum pressure in the working process, but the smaller the mean indicated piston pressure and hence the indicated horsepower of the engine.

Table III

Variation of Characteristic Quantities for Different Parts of the Variable Combustion Period at n = 1000 R.P.M.

$\frac{z}{z_v}$	0	0.25	0.50	0.75	1
		$\alpha_v = 15^\circ$,	$z_v = 0.0025$ sec.		
α ....degrees	0	3.75	7.50	11.25	15.0
ξ ...... %	0	0.1	0.5	1	1.7
p ... at abs.	30	36.2	46	56.5	72.1
t .... °C	598	892	1239	1665	2326
		$\alpha_v = 30^\circ$,	$z_v = 0.0050$ sec.		
α ....degrees	0	7.5	15	22.5	30
ξ ...... %	0	0.45	1.7	3.8	6.7
p ... at abs.	30	34.7	40	42.6	47.6
t .... °C	598	880	1209	1584	2203
		$\alpha_v = 45^\circ$,	$z_v = 0.0075$ sec.		
α ....degrees	0	11.25	22.5	33.75	45
ξ ...... %	0	0.9	3.8	8.5	14.7
p ... at abs.	30	32.4	34.6	31.5	31.4
t .... °C	598	858	1245	1570	2141
		$\alpha_v = 60^\circ$,	$z_v = 0.0100$ sec.		
α ....degrees	0	15	30	45	60
ξ ...... %	0	1.7	6.7	14.7	25
p ... at abs.	30	29.6	25.1	20.7	20
t .... °C	598	836	1065	1356	1918

If we calculate the mean indicated piston pressure p_i in atmospheres from the indicator diagrams, the indicated horsepower per liter stroke volume, is then

$$\frac{N_i}{V_h} = \frac{p_i\ n}{900} [HP/l].$$

When computed for an hour, the work done is

$$AL = 632\ \frac{N_i}{V_h} = 0.702\ p_i\ n\ [kcal/l\,h]$$

Since

$$B_l = \frac{288\ p_1}{10^3\ \lambda\ L_{min}\ T_1}\ [kg/l]$$

fuel is burned by one liter of air under the conditions at the close of the intake (p_1, T_1) with an air excess λ, the chemical energy, added hourly in the form of fuel, is

$$Q = 30\ n\ B_l\ h_u$$

$$= 8.64\ \frac{p_1\ h_u\ n}{T_1\ \lambda\ L_{min}}\ [kcal/l\,h]$$

and the efficiency of the energy transformation for the engine (heat losses by radiation and conduction being neglected) is

$$\eta = \frac{AL}{Q}$$

$$= 0.0813\ \frac{T_1\ \lambda\ L_{min}}{p_1\ h_u}\ p_i.$$

With $p_1 = 1$ atm. abs., $t_1 = 57^\circ C$ and $\lambda = 1.50$, $\eta = 0.0486\ p_i$ for the particular gas oil used. Table IV and Figure 9 show the dependence of the individual factors on the

length of the combustion period. The power output per liter N_i/V_h, the efficiency η and the ratio of pressure increase p_{max}/p_2 diminish considerably with diminishing combustion speed. The latter is inversely proportional to the combustion period, $c = k/z_v$ (m/sec.). For $c = \infty$, the limiting values are $\eta = 0.590$ and $p_{max}/p_2 = 3.25$.

Table IV.

Effect of the length of the Combustion Period at n = 1000 R.P.M.

a_v degrees z_v s	0 0	15 0.0025	30 0.0050	45 0.0075	60 0.0100
$\frac{1}{z_v}$ ...... s^{-1}	∞	400	200	133.3	100
p_i .. at abs.	12.1	11.3	10.4	9.7	8.5
$\frac{N_i}{V_h}$ .... HP./l	13.45	12.56	11.57	10.79	9.45
AL ...kcal/l h	8500	7930	7300	6810	5970
Q ...... "	14410	14410	14410	14410	14410
η	0.590	0.550	0.506	0.473	0.414
$\frac{p_{max}}{p_2}$	3.25	2.40	1.59	1.15	1

The representation of η and p_{max}/p_2, as plotted against the combustion speed at constant R.P.M. in Figure 10, shows that the combustion speed must have a certain value, in order that the energy utilization in the engine may not be too far below the limiting value. As will be shown by the following experimental results, the formation of the mixture and the correctly

timed combustion of heavy oils in compressorless engines now offer no insurmountable difficulties, even at very high rotative speeds. The limit seems rather to lie in the fact that, above a certain speed, it is no longer possible to obtain high mean indicated pressures, without employing special devices for supplying the cylinder with sufficient charging air.

With respect to reasonable fuel consumption, we can now assume, as this limit, n = about 1000 R.P.M. and a the final compression pressure of 30 atm. abs. For the chosen example, the specific fuel consumption of the engine, neglecting heat losses due to radiation and conduction, computed for a thermal value of 10,000 kcal/kg, is then

$$B_i = \frac{30\ n\ B_l}{N_i/V_h}\ \frac{h_u}{10000}\ [\text{kg/HP}_i\text{/h}] .$$

Table V shows that B_i lies, according to the combustion speed, between 0.153 and 0.107 kg/HP$_i$/h. It is worthy of note that the small consumption values can be obtained only at the expense of a greater pressure increase in the combustion. An infinitely great combustion speed corresponds to the limiting value $B_i = 0.107$ kg/HP$_i$/h. Fortunately, however, the consumption B_i increases relatively slowly with decreasing combustion speed, while the maximum pressure falls rapidly. The efficiency η of the energy transformation for medium combustion speeds may therefore be considered as very favorable.

Table V.

Effect of Combustion Speed on Fuel Consumption and Efficiency at $n = 1000$ R.P.M.

$\frac{1}{z_v}$ ...... s^{-1}	100	133.3	200	400	∞
p_{max} at abs.	30	34.6	47.6	72.1	97.8
t_{max} ...... °C	1918	2141	2203	2326	2450
B_i .... kg/HP_i/h	0.153	0.134	0.125	0.115	0.107
B_e .... kg/HP_e/h	0.207	0.184	0.174	0.162	0.153
η_i	0.414	0.472	0.506	0.545	0.590
η_e	0.302	0.341	0.360	0.387	0.410
η_m	0.74	0.73	0.72	0.71	0.70

The above considerations also render it possible to determine how far we can reduce the specific fuel consumption B_e (kg/HP_e/h) per brake horsepower. For this purpose, the mechanical efficiency η_m is assumed to be between 0.70 and 0.74, according to the maximum pressure. We obtain $B_e = 0.153$ kg/HP_e/h as the limiting value for $c = \infty$. With decreasing combustion speed, B_e increases up to 0.207 kg/HP_e/h.

Since well-designed high-speed engines, as shown by the following experimental results yield fuel-consumption figures which fall between these two values, it follows that mechanical injection has already justified itself, even for high-speed Diesel engines for vehicles.

Experiments with the Dorner Engine

Thorough tests of compressorless high-speed Diesel engines for vehicles have not hitherto been made. The few data from testing-bench experiments are inadequate for a complete presentation of this new engine type under various operating conditions. For this reason, the results of the investigation of the four-cylinder Dorner oil engine (in which all the newest testing devices were used), merit attention.

The four-stroke-cycle engine (Figs. 11-12) has four cylinders of 95 mm (3.74 in.) bore and 160 mm (6.3 in.) stroke, with mechanical fuel injection. The effective horsepower is 35 at 1000 R.P.M. The fuel was gas oil with a lower heating value of 10,100 kcal/kg and a specific gravity of 0.851 at 15°C. Theoretically its combustion required 12.11 m^3 of air (at 10°C and 760 mm Hg) for 1 kg of fuel.

The engine is built by Max Jüdel, Stahmer and Bruchsal at Osnabrück. On the two-part crank case there are two blocks, each containing two cylinders. Both have removable cylinder heads (Fig. 13) with vertical intake and exhaust valves and two separately attached fuel pumps connected with injection nozzles for each cylinder. The nozzles can be easily removed by loosening one nut. The valves and fuel pumps are actuated by two cam shafts in the upper part of the crank case, which are driven by the crank shaft by means of chains. The oil pump is located in

the lower part of the crank case, which serves as a trough for collecting the oil. Both the oil pump and the cooling-water pump are driven by a vertical shaft, which, in turn, is operated through spiral gears, by the valve-drive cam shaft. The oil flows through a filter to the pump. On the cam shaft there is a centrifugal governor, which acts as an over-speed governor and does not allow the engine to exceed 1200 R.P.M. The engine can be adjusted, while running, to any desired R.P.M. between 400 and 1000.

The engine housing is perfectly dustproof. The fuel pump and nozzles (Fig. 14-15) merit special notice. The fuel pump has two discharge valves v. Instead of a suction valve, there is a suction port s, through which the fuel flows to the pump chamber when it is opened by the plunger p. The latter has, on its lower end, a spring-loaded guide piston e, in which the push-rod l rests. The fuel cam n, by means of the roller r_1, actuates the cam lever h, around the pivot O, which in turn transmits the cam motion through the roller r_2 to the push rod l and consequently to the plunger p of the fuel pump.

The amount of fuel to be injected into the cylinder is regulated by altering the pump stroke. The governor turns the governor shaft w, which acts, by means of the guide link t, on the push rod l in such a manner that, for every loading of the engine, the push rod l is given a certain definite position by the radius ρ, so that the greatest fuel charge corresponds to

the maximum value of ρ and the rest position of the plunger corresponds to the minimum value of ρ. The fuel delivered by the pump passes through the discharge valves v into the nozzle body d (Fig. 15) and, after leaving the nozzle at k, is sprayed into the cylinder in a conical jet.

In the experiments, the effective horsepower was determined by a Junkers water brake, the R.P.M. by a revolution counter, the fuel consumption by weighing, and the air used was measured by a gas meter. Mercury thermometers showed the inlet and discharge temperatures of the cylinder-cooling water, its quantity being measured by means of calibrated tanks. Gas samples were continually taken from all the cylinders. Two series of experiments were conducted. In the first series at normal R.P.M., the load was altered. In the second series the R.P.M. was varied, in order to obtain the especially important efficiency curve for high-speed engines. Every experiment was continued at least an hour, in order to permit the engine to reach a stable condition before any readings were made.

Experimental results.- These are shown in Table VI and Figures 16-18. The mechanical efficiency was approximated from the heat balance. Disregarding the losses through radiation and imperfect combustion, the heat q_r (Fig. 18) corresponds to the work of friction, and the relative heat value of the indicated work is approximately $q_i \sim q_e + q_{r_{\alpha=1.15}}$. Consequently,

$\eta_m = \frac{q_e}{q_e + q_{r_{\alpha=1.15}}}$. It is here further assumed that the work of friction is independent of the load.

The variation in the fuel consumption B_e (kg/HP$_e$/h), with change of load for high-speed engines, is similar to that for other compressorless engines. The specific oil consumption remains almost constant over a long range of load. From $\alpha = 1$ to $\alpha = 0.5$, the consumption increases only about 13%. The exhaust temperatures t_z ($^\circ$C) are uniformly low, and the mean effective piston pressure p_e (atm.) is high. With 15% air excess during the endurance test, the engine gave $p_e = 7$ atm. with an output of $N_e/N_h = 7.6$ HP$_e$/liter, and $\eta_{t_e} = \frac{632\ N_e}{B\ h_u} 100 =$ 30.1% of the heat energy of the fuel was converted into useful work on the crank shaft.

Table VI.

Results of Experiments with the Dorner Engine.

Experiment Number	1	2	3	4
1. Constant R.P.M.	Variable load.			
Barometer, b_{15} .... mm Hg	758	757	757	756
Brake load, P .... kg	35.3	19.8	9.9	0.48
Rotative speed, n .... R.P.M.	976	1,050	1,064	1,053
Effective horsepower, N_e ... HP.	34.5	20.8	10.5	0.50
Load factor, α	1.15	0.694	0.352	0.017
Mean effective pressure, p_e atm.	7.00	3.93	1.96	0.090
Total fuel consumption, B .... kg/h	7.20	4.490	2.863	1.780
Specific fuel consumption, B_e .... $kg/HP_e/h$	0.209	0.216	0.273	3.56
Air consumption (red.) L_r .... m^3/h	105.6	113.5	117	115.2
Air-fuel ratio, L_r/B ..... m^3/kg	14.67	25.27	40.8	64.7
Air excess, λ	1.21	2.09	3.37	5.34
Volumetric efficiency, $\eta\lambda$	0.795	0.795	0.808	0.804
Exhaust temperature, t_z ..... °C	569	405	259	133
Exhaust quantity, m" .... mol/kg	0.632	1.050	1.708	2.685
Mean mol. heat of exhaust gases, $(\mu c_p")_{t_o}^{t_z}$ kcal mol°C	7.69	7.56	7.12	7.05
Exhaust heat loss, Q_z ... kcal/h	19,240	11,620	8,350	4,170
Temp. of water inlet, t_e .... °C	16	16	17	18
" " " discharge, t_a °C	57	79	72	71
Quantity of water, W ...... kg/h	545	174.6	146.1	80.8

Table VI (Cont.)

Results of Experiments with the Dorner Engine

Experiment Number	1	2	3	4
1. Constant R.P.M.	Variable load.			
Heat loss to water, Q_k...kcal/h	22,350	11,000	8,190	4,280
Torque, M_d .... mkg	25.3	14.2	7.1	0.34
Thermal efficiency, η_{t_e}	0.300	0.290	0.230	0.017
Heat Balance:				
Useful work in heat units, Q_e .... kcal/h	21,860	13,140	6,650	310.000
Cooling-water heat loss, Q_k .... kcal/h	22,350	11,000	8,190	4,280
Exhaust heat loss, Q_z .... "	19,240	11,220	8,350	4,170
Friction and radiation, Q_r "	9,350	9,590	5,760	9,230
Heat content of fuel consumed, Bh_u .... kcal/h	72,800	45,350	28,950	17,990
q_e	0.300	0.290	0.230	0.017
Heat losses in fractions of the heat consumed: q_k	0.307	0.243	0.284	0.238
Heat losses in fractions of the heat consumed: q_z	0.264	0.256	0.288	0.232
q_r	0.129	0.211	0.198	0.513

Table VI (Cont.)

Results of Experiments with the Dorner Engine

Experiment Number	5	6	7	8
2. Variable R.P.M.				
Brake load, P .... kg	26.30	33.84	36.20	35.40
Rotative speed, n ...... R.P.M.	1091	887	721	471
Effective horsepower, N_e ... HP.	28.7	30	26.1	16.7
Mean effective pressure, p_e .... atm.	5.21	6.71	7.18	7.02
Torque, M_d .... mkg	18.8	24.2	25.9	25.4
Total fuel consumption, B, ... kg/h	5.94	6.19	5.73	3.82
Spec. fuel consumption, B_e .... kg/HP$_e$/h	0.207	0.206	0.220	0.229
Thermal efficiency, η_{t_e}	0.303	0.304	0.286	0.274

From the air measurement it follows that, at $n = 1000$ R.P.M., like volumes of air ($L_r = 108.9$ m³/h at 10°C and 760 mm Hg) are drawn in for all loads. Consequently the volumetric efficiency of the cylinders ($\eta_\lambda = 0.80$) is also constant. Since the engine was adjusted to the load by varying the amount of fuel injected, the air excess λ increases considerably toward the idling speed.

For determining the thermal conditions, especially of the course of the combustion, it is of advantage to plot the specific fuel consumption B_i (kg/HP$_i$/h), the heat value of the mixture $Q_m = \frac{h_u}{1 + \lambda\ L_{min}}$ (kcal/kg), the air excess λ and the

exhaust temperature t_z (°C) against the mean indicated piston pressure p_i (atm.) (Table VII and Fig. 19). It is seen that $B_i = 0.146$ kg/HP$_i$/h, quite in contrast with a carburetor engine, is constant over a great load range ($\alpha = 1.15 - 0.3$). Hence, we must conclude that, although the air excess increases considerably with diminishing load and consequently the heat value of the mixture constantly becomes smaller, the ignition and combustion of the heavy oil take place rapidly and surely in the short periods of time available. First at 3.5 fold air excess, corresponding to a heat value of the mixture of 194 kcal/kg, B_i increases as a result of the slow combustion (Figs. 19-20).

Table VII.

Effect of Mean Indicated Piston Pressure on the Working Process at Constant Speed (n = 1000 R.P.M.)

α	0.017	0.352	0.694	1	1.15
p_e .... at	0.090	1.96	3.93	6	7
η_m	0.20	0.63	0.69	0.70	0.70
p_i .... at	0.45	3.11	5.70	8.58	10
B_i .. kg/HP$_i$/h	0.712	0.172	0.145	0.145	0.146
λ	5.34	3.37	2.09	1.50	1.21
Q_m ....kcal/kg	128	201	320	440	540

A Diesel engine cannot function properly with an insufficient supply of air $(\lambda < 1)$, because the combustion is then incomplete and very smoky. While carburetor engines attain their maximum power as a result of the dissociation of the water vapor and carbon dioxide formed by the combustion, although at the expense of the fuel consumption, a compressorless mechanical injection engine generates its maximum power at a very small excess of air $(\lambda \sim 1.2)$. The heat value of the mixture here reaches almost the limiting value $Q_{max} = \frac{h_u}{1 + L_{min}} = 646$ kcal/kg.

As shown by Figs 19-20, the engine functions at the normal power $(\alpha = 1,\ p_i = 8.6$ atm.) with 50% air excess and a heat value of the mixture of 440 k cal/kg, while it readily reaches a mean piston pressure of $p_i = 10$ atm. at 15% overload, with only 21% excess air and a heat value of the mixture of 540 kcal/kg. The limit lies at $p_i = 11$ atm. with respect to the charging air of the cylinder (Volumetric efficiency $\eta_\lambda = 0.80$). Here the heat value of the mixture would have the maximum value $Q_{max} = 646$ kcal/kg.

This limit can be exceeded only by increasing η_λ. If we could obtain $\eta_\lambda = 1$ (perhaps by some special method of charging), then the mean indicated piston pressure for a four-stroke engine would reach the limiting value

$$p_i = \frac{27\ \eta_\lambda}{B_i\ \lambda\ L_{min}} = 13.7 \text{ atm.}$$

at 10% air excess and perfect combustion. The heat value of the

mixture would then increase to 591 kcal/kg and remain only 8.5% below its maximum value.

From the above considerations, it follows that the transformation speed of the chemical energy of the heavy oils into heat with suitable atomization of the fuel, even in the short combustion periods of high-speed engines, is always adequate, so that, in this respect, the light oils have no advantage over the heavy oils. The limit for the maximum efficiency of a given cylinder volume is not, therefore, in the combustion speed, but in the rotative speed above which, without a special charging method, no adequate amount of air can be introduced into the cylinder.

A high-speed Diesel engine has an important advantage over a carburetor engine, in that the formation of the mixture is independent of the degree of air excess (Fig. 20). From idling speed to overload, in every instance the combustion was excellent and the exhaust clear, an indication that the combustion was proceeding properly, provided only that λ was above unity, without regard to whether this value was exceeded but slightly ($\lambda = 1.21$) or very greatly ($\lambda = 5.34$).

These results were confirmed by the remarkably low losses evidenced by the functioning of the engine. In the numerical determination, the thermodynamic efficiency ($\eta_{thermod.}$) was determined for the normal power ($\alpha = 1$) at the normal revolution speed ($n = 1000$ R.P.M.). For this purpose (Fig. 16),

$\lambda = 1.50$, $\eta_m = 0.70$, $p_e = 6$ atm., $p_i = 8.6$ atm. If $p_i{}^o$ = the mean indicated pressure of the engine, then $\eta_{thermodynamic} = p_i/p_i{}^o$. $p_i{}^o$ can, however, be derived from the PV diagrams of the engine (Fig. 8) developed for the abovementioned conditions. It is necessary to have an accurate knowledge of the crank angle α, at which the combustion ends in the engine. The magnitude of the thermodynamic efficiency $\eta_{thermod.}$ and of the relative losses $\zeta = 1 - \eta_{thermod.}$ depends, therefore, on the combustion speed. The corresponding values are as follows:

α_v	$p_i{}^o$	$\eta_{thermod.}$	ζ
0^o	12.1 atm.	0.71	0.29
15^o	11.3 "	0.76	0.24
30^o	10.4 "	0.83	0.17

On the basis of the low exhaust temperatures taken in the test, the engine must have worked with a large expansion ratio of the burned charge and α_v must consequently have been small. For $\alpha_v = 12^o$, $\zeta = 0.25$, $\eta_{thermod.} = 0.75$, and $p_i{}^o = 11.4$ atm. The time, within which the pressure rises from the final compression pressure of 30 atm. abs. to its maximum value of 61 atm. abs., is

$$z_v = \frac{60}{n} \frac{\alpha_v}{360} = 0.002 \text{ sec.}$$

From Figure 9 it follows, for $z_v = 0.002$ sec., that, as the efficiency of the engine (heat losses being neglected) $\eta_v = 0.56$.

The thermal efficiency of the engine is therefore $\eta_{t_i} = \eta_v\ \eta_{thermod.} = 0.56 \times 0.75 = 0.42$ relative to the effective power $\eta_{t_e} = \eta_{t_i}\ \eta_m = 0.42 \times 0.70 = 0.294$, and the relative losses are $\zeta = 0.25$. Although, for the lack of an accurate indicator diagram, α_v cannot be given exactly, this consideration shows that the total working losses of the tested engine, at full load and normal revolution speed, constitute about a quarter of the theoretically possible work. If, on this basis, the high-speed Diesel engine is compared with the stationary air-injection engine previously tested by me (V.D.I. 1923, p. 282), it is found that in spite of the great difference in the rotative speeds and in the working cycle, the losses of the two engines are the same.

This fact best explains the great progress which the Diesel engine has made, through the development of the compressorless type, toward becoming a high-speed vehicle engine. The characteristic curves of the high-speed Diesel engine (Table VIII, Figs. 17 and 21), effective power and torque, plotted against the R.P.M., $N_e = f(n)$ and $M_d = \varphi(n)$, show that the maximum power, at the normal speed $n = 1000$ R.P.M., is reached, up to which N_e increases proportionally with n. An important advantage over the carburetor engine consists in the fact that the formation of the mixture is independent of the rotative speed. In contrast with gasoline engines, the Diesel engine requires no external device for adapting the mixture and the com-

bustion to the variations in the operating conditions.

With increasing speed, several other noteworthy phenomena occur in the high-speed Diesel engine (Fig. 21). Although the volumetric efficiency η_λ decreases with increasing speed, the air excess λ increases up to the normal speed and consequently, the heat value of the mixture $\frac{h_u}{1 + \lambda\ L_{min}}$ decreases. Although the quantity drawn in during each working cycle decreases, the specific amount of air $\frac{L}{B} = \lambda\ L_{min}$ increases, since, with increasing speed, the turbulence in the cylinder increases and the combustion speed is thereby considerably increased, which results in the better utilization of the oxygen of the air charge.

The increase of the specific fuel consumption B_e(kg/HP$_e$/h), from its minimum value at the normal revolution speed in the direction $n < 1000$ R.P.M., is due to the combustion becoming more and more imperfect, as the result of the smaller turbulence in the combustion chamber, while the increase in the fuel consumption in the direction $n > 1000$ R.P.M., is due to the rapid approach to the theoretical quantity of air, $\lambda = 1$. The flat course of $B_e = f(n)$, (Fig. 17) shows, however, that the specific fuel consumption was only slightly affected by the speed. At all speeds there was an excess of air. Consequently, the heat value of the mixture was always below $Q_{max} = 646$ kcal/kg, and ignition was always sure, even at the lowest speed.

The working cycle of the high-speed Diesel engine is therefore adapted to the economical combustion of heavy oils in high-

speed vehicle engines. The structural progress already made encourage us to hope that the time is not far off, when this new type of engine will come into general use on vehicles.

Table VIII.

Characteristics of the High-Speed Diesel Engine

n .... R.P.M.	471	721	887	1000	1091
η_λ	0.92	0.88	0.84	0.80	0.72
L .... kg/h	67.4	104.1	127.1	131.5	126.9
λ	1.21	1.25	1.40	1.50	1.46
Q_m .... kcal	542	526	470	441	452
$\frac{N_e}{4\ V_h}$ .. HP/liter	3.69	5.78	6.61	6.74	6.33

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

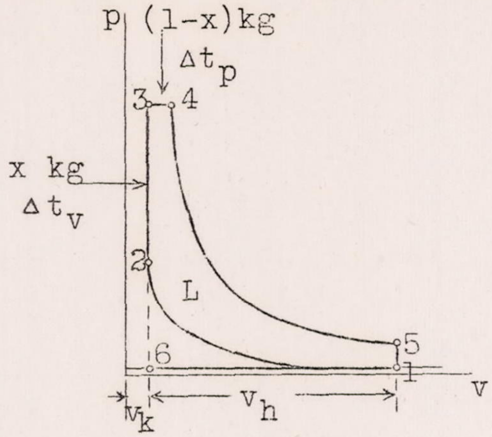


Fig.1 Theoretical indicator diagram of a Diesel engine.

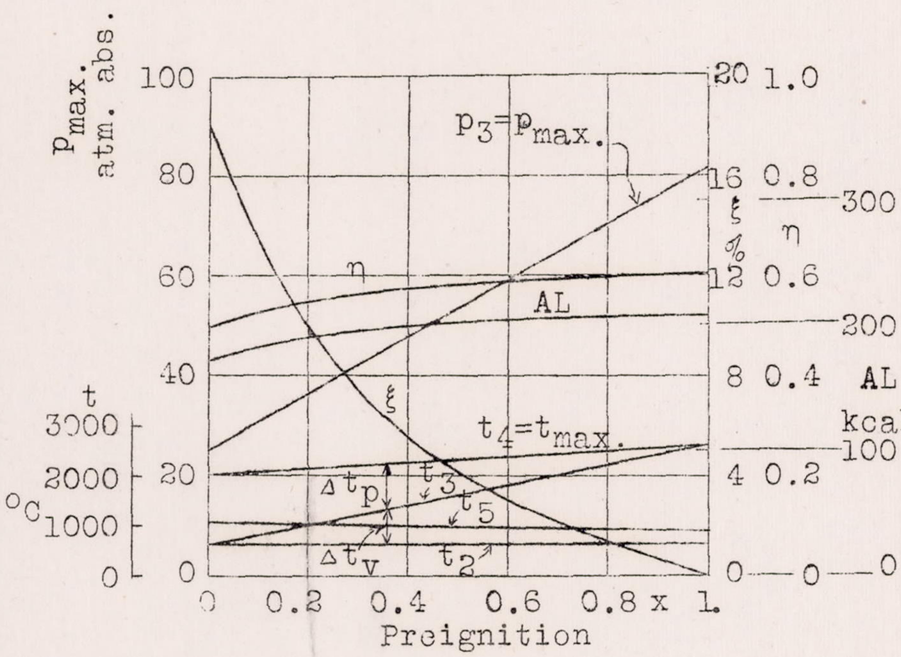


Fig.2 Effect of the preignition x on the working process.

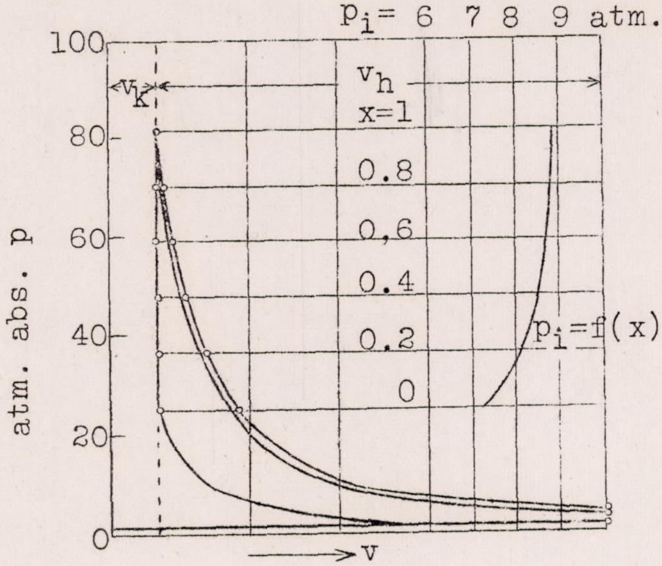


Fig.3 Indicator diagrams for various preignitions x. Mean indicated pressure p_i plotted against the course of the combustion.

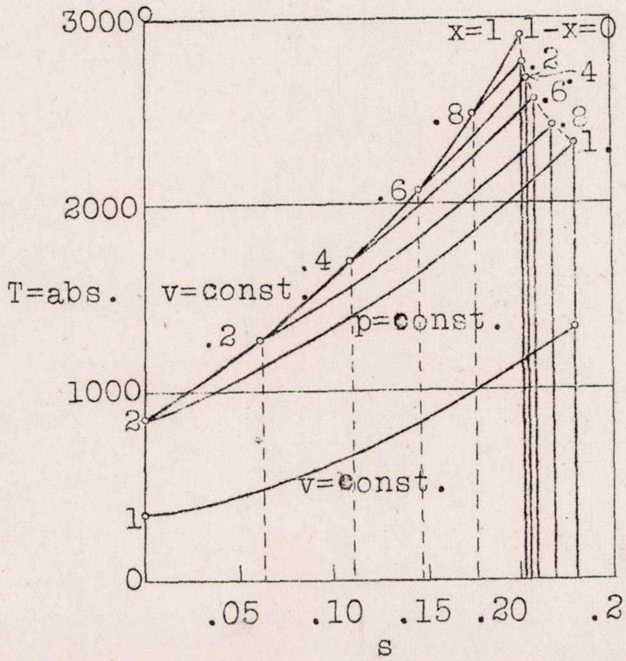


Fig.4 Ts diagram for various degrees of preignition x.

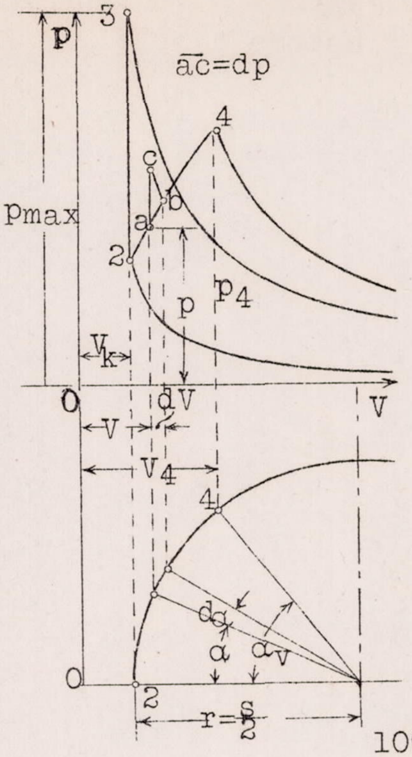


Fig.5

Calculation of the combustion line 2-4 in the pV diagram.

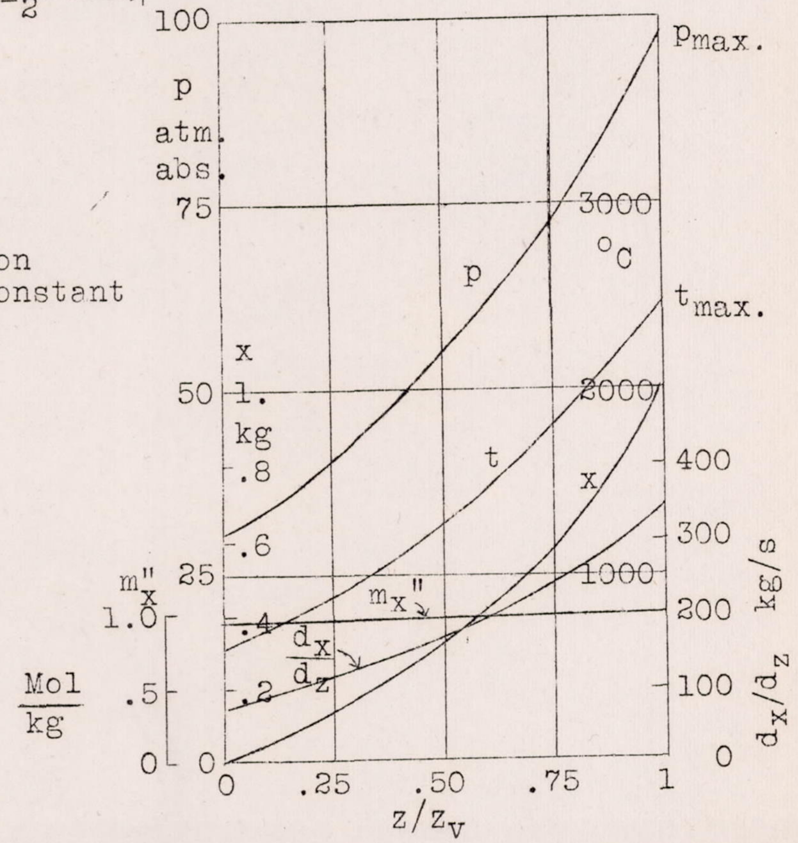


Fig.6

Effect of time on combustion at constant volume.

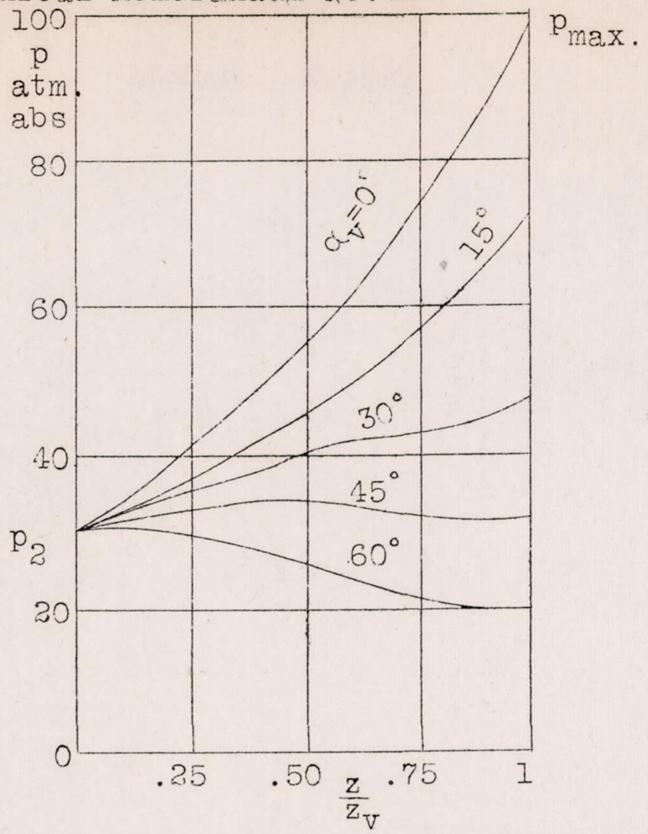


Fig.7 Pressure plotted against combustion speed.
Combustion time $z_v = \alpha_v / 6n$ (sec).

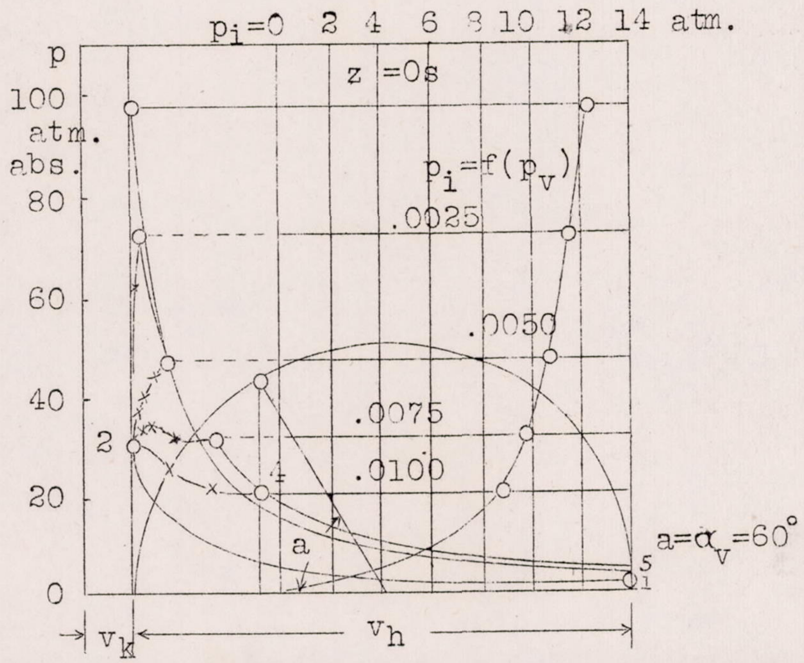


Fig.8 Effect of combustion speed on indicator diagram and on the mean indicated piston pressure at constant n=1000 R.P.M. The combustion time $z_v=\alpha_v/6n$(sec).

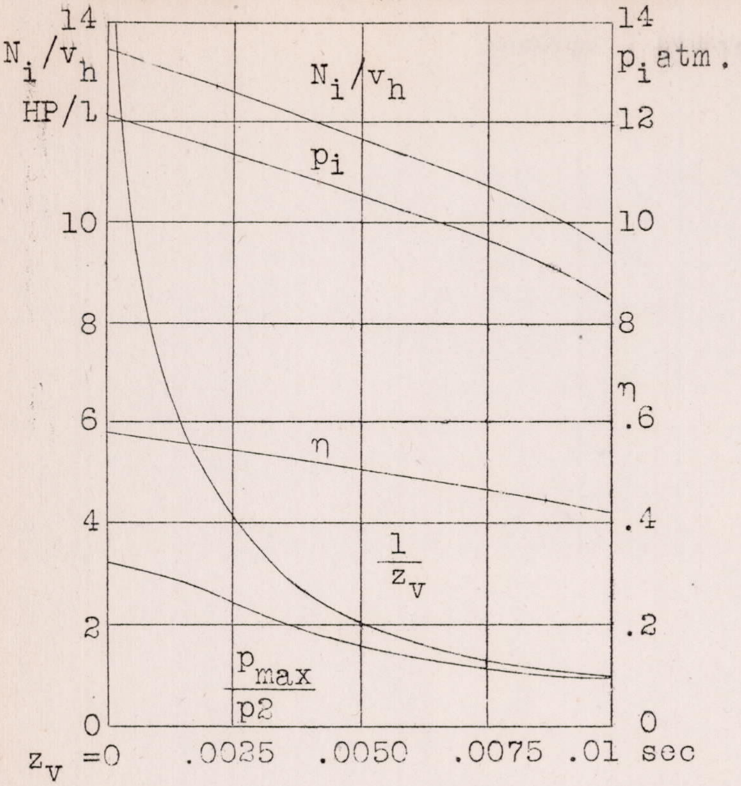


Fig.9 Effect of combustion time on the indicated engine power. Combustion speed proportional to $1/z_v$

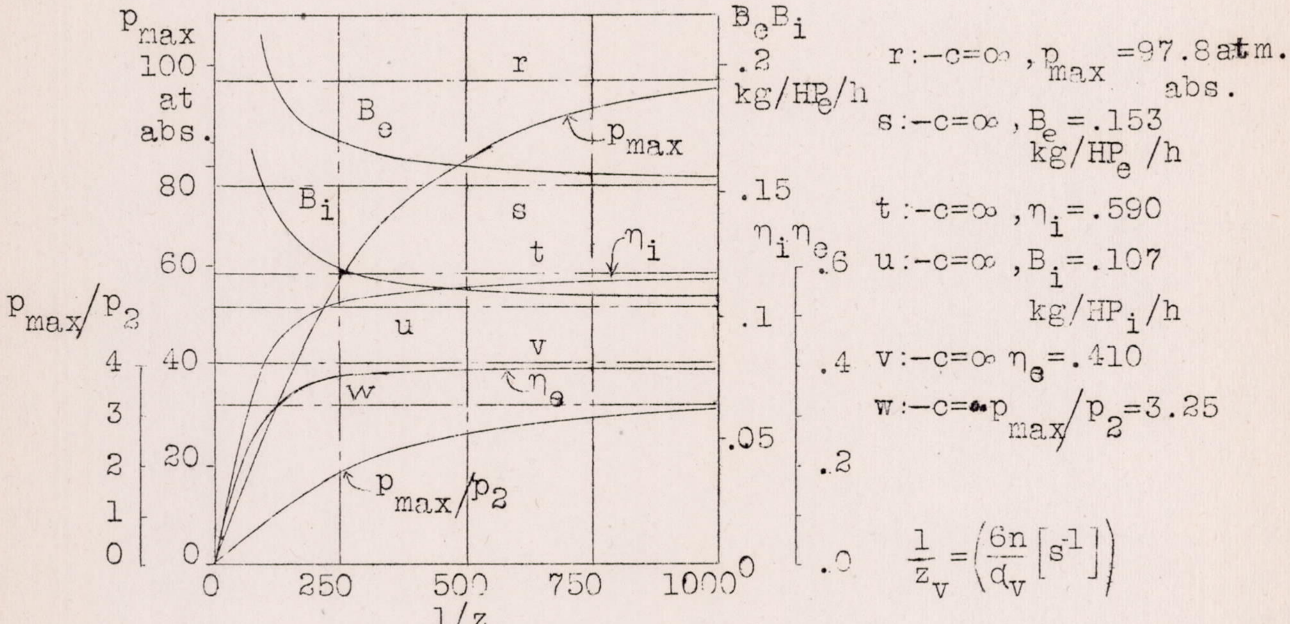


Fig.10 Effect of combustion speed on the efficiency, specific fuel consumption and pressure increase during combustion.

Figs. 11 & 12

Dorner 30 HP_e heavy-oil engine

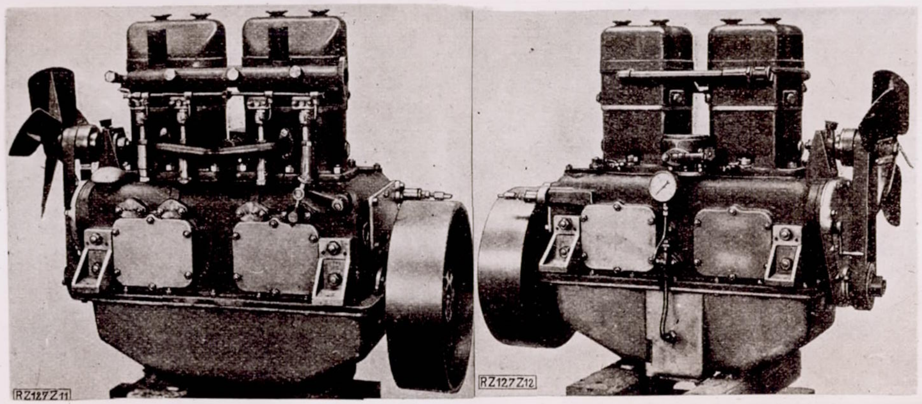
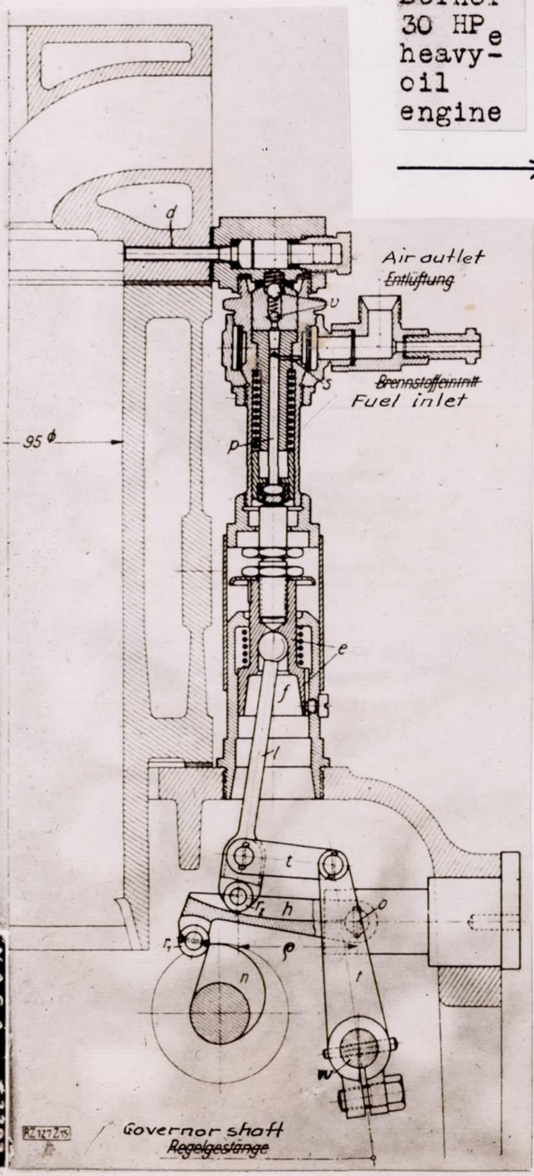
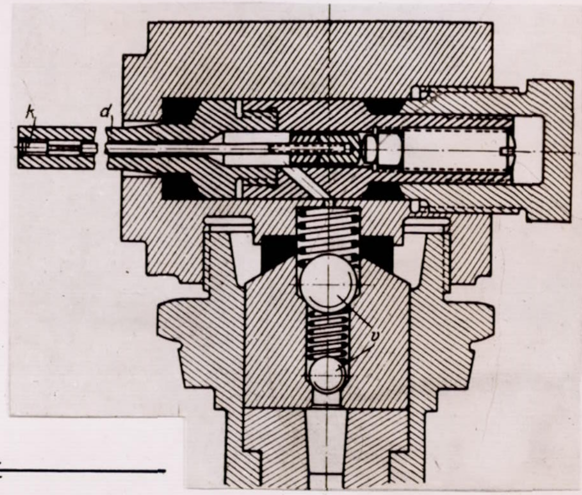
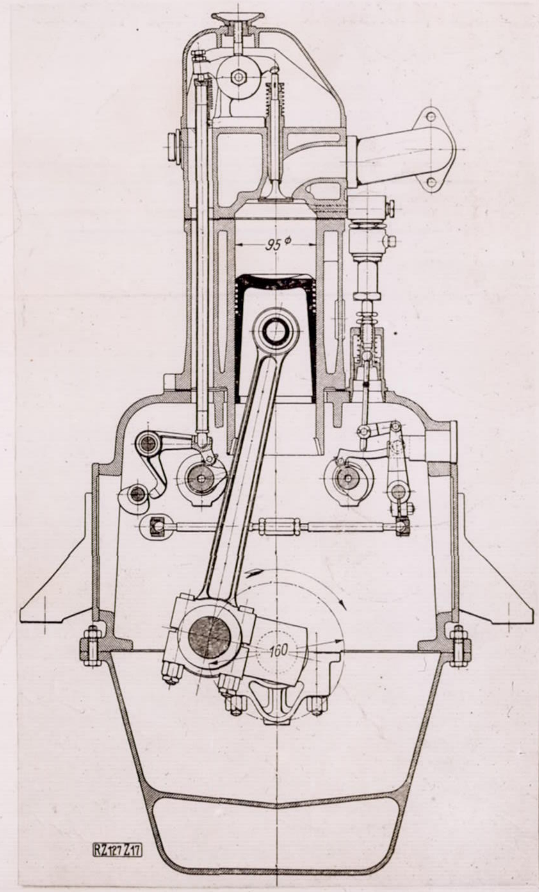


Fig.13 Dorner, 4 cyl, four-stroke-cycle-engine.
n - 1000 R.P.M.
Bore - 95 mm (3.74") Stroke - 160 mm (6.3").







Figs. 14 & 15 Fuel pump and enlarged section of nozzle.
v, discharge valve; s, suction port; p, pump plunger; e, guide piston; l, push-rod; n, fuel cam; r_1, roller; o, pivot; h, cam lever; r_2, roller; w, governor shaft; t, guide link; ρ, radius; d, nozzle body; k, nozzle tip.

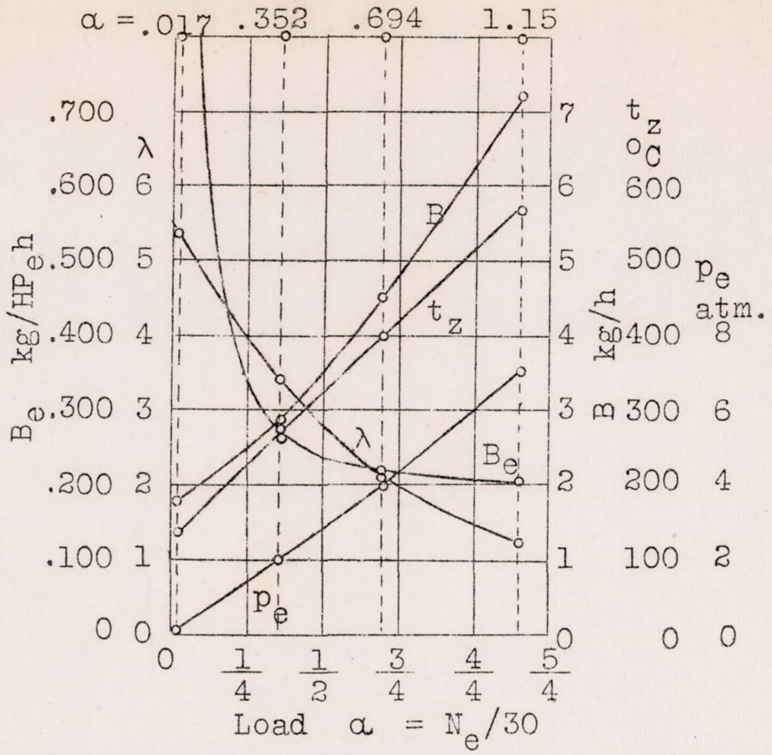


Fig.16 Fuel consumption B (total) and B_e (specified), mean piston pressure p_e, exhaust temperature t_z, air excess λ, plotted against load at constant n = 1000 R.P.M.

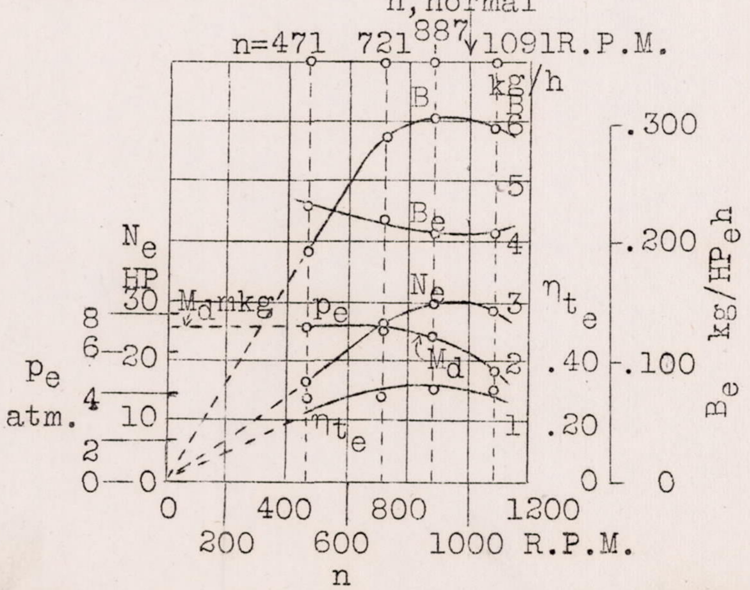


Fig.17 Effective power N_e, fuel consumption B and B_e, piston pressure p_e, torque M_d, thermal efficiency η_{te}, plotted against R.P.M.

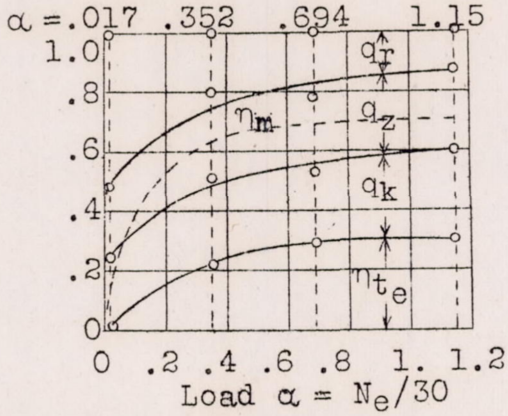


Fig.18 Heat distribution plotted against load at constant n = 1000 R.P.M.

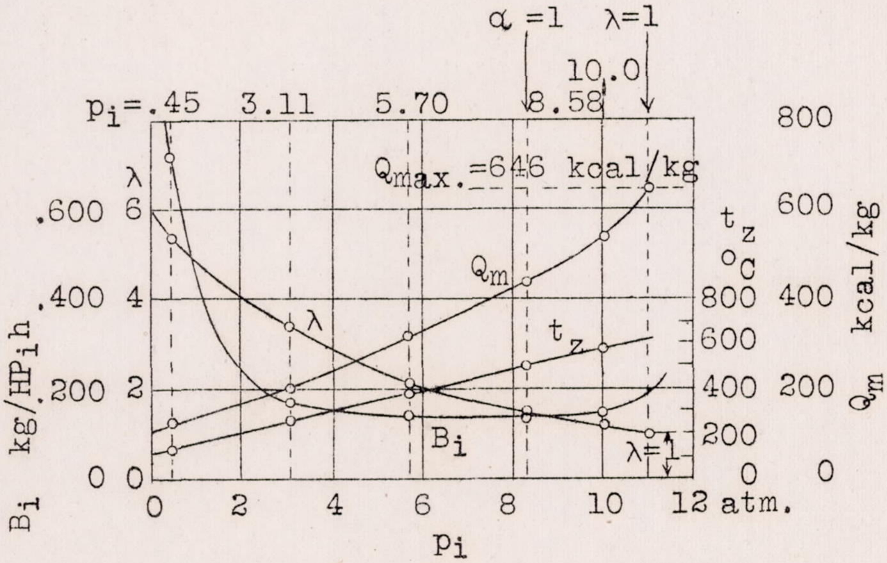


Fig.19 Specific fuel consumption B_i (per I.H P.), air excess λ, heat value of mixture Q_m, exhaust temperature t_z, plotted against mean indicated pressure at constant n = 1000 R.P.M.

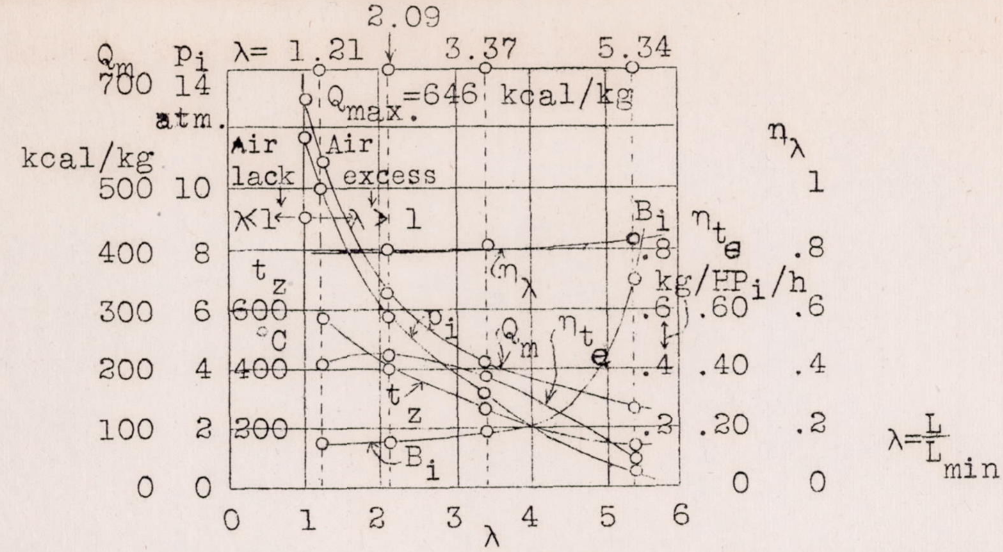


Fig.20 Mean pressure p_i, fuel consumption B_i, volumetric efficiency η_λ, heat value of mixture Q_m, exhaust tempature t_z, plotted against air excess at constant n=1000 R.P.M. Thermal efficiency η_{t_e}.

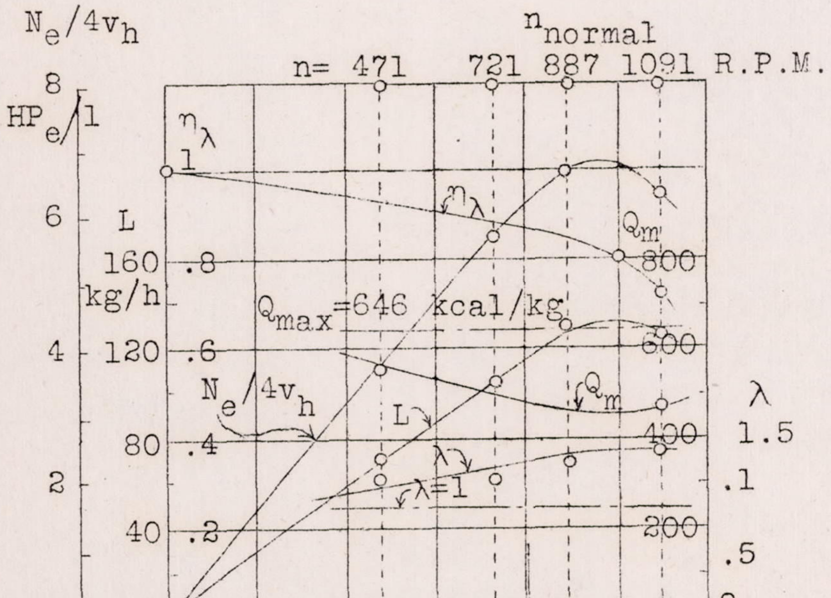


Fig.21 Power output per liter, amount of air L, volumetric efficiency η_λ, heat value of mixture Q_m, air excess λ, plotted against the R.P.M.